Chilled Water Feeder by using Dynamic Ice in a Dairy Product Plant

Daisuke Mito [1,*], Tatsunori Mano [1], Masayuki Tanino[1],
Masaru Hongo[2], Kazuo Wakasa[2], Koji Matsumoto[3]

[1] Takasago thermal engineering Co., Ltd. R&D Center, Kanagawa, Japan
[2] Takasago thermal engineering Co., Ltd. Sapporo Brunch, Sapporo, Japan
[3]Department of Precision Mechanics Chuo University, Japan

Abstract: In Japan, a supercooling-type ice storage system has been introduced for buildings air-conditioning and district heating and cooling. Ice slurry, which is the thermal storage material of the system, has an excellent melting characteristic, and is suitable for low-temperature chilled water supply for the food industry. We developed an operation control technology based on the use conditions, which are different from those for air-conditioning and unique to food factories, and delivered an ice storage system as a chilled water feeder for a dairy product plant.

This paper presents the design calculation results for determining the refrigerator and storage tank capacity and the ice packing factor of the thermal storage tank, which is needed to maintain the chilled water temperature. Furthermore, we show the result of the application of new technologies such as the control in a fully accumulated state and of the water supply for the actual load processing.

Keywords: Thermal storage, Chilled water, Food industry, Ice packing factor.

1. Introduction

We introduced a large-scale dynamic-type ice storage system[1,2] where ice exhibiting the supercooling phenomenon of water is made, to practical use for the first time in Japan in 1988. We have achieved many results related to the introduction of this system as a thermal storage system to provide electric-load leveling for district heating and cooling equipment and in buildings used for different purposes, including office buildings, university facilities, business spaces, hospitals, plants, and others. In 2004, we commercialized the system, which included drastically downsizing the ice making section and improving its operating efficiency[3-5]. In addition, we developed a design tool to predict the change in the supply water temperature in the melting process[6].

Because ice slurry, which is the thermal storage material of the system, has an excellent melting characteristic and is safe fresh water that does not include additives, it is also suitable for low-temperature chilled water supply for the food industry. Thus, we recently developed operation control technology based on the use conditions for this industry, which are different from those for air-conditioning and unique to food factories[7], and delivered an ice storage system as a chilled water feeder to a dairy product plant.

In this report, in Chapter 2, we will first explain the basic design for stably supplying chilled water, and the specific operation control technology for a chilled water feeder. In Chapter 3, we will explain the outline of the chilled water feeder that was introduced to the actual plant, while Chapter 4 discusses its performance.

Symbol

c_l	specific heat of water	$(\mathrm{kJ \cdot kg^{-1} \cdot K^{-1}})$
h_L	latent heat of ice	$(\mathrm{kJ \cdot kg^{-1}})$
Δh_w	water level difference from normal water level	(m)
IPF	ice packing factor	(%)

ISBN 978 979 8510 61 8

N	number of water exchanges in water region in the tank	(number h^{-1})
P	arrangement pitch of jet nozzle	(m)
q	cold heat quantity of melting ice in ice layer	(kW)
q_{load}	thermal load	(kW)
q_{ref}	cooling capacity of an ice generator	(kW)
T_b	temperature of water flowing out of ice layer	(K)
T_{in}	influent temperature of water for calculation of ice melting	(K)
T_{melt}	melting point of ice	(K)
T_{out}	outlet temperature of chilled water (intake)	K (°C)
U_c	flow rate of chilled water in ice slurry	($m^3 \cdot h^{-1}$)
U_m	flow rate of water jet	($m^3 \cdot h^{-1}$)
u_j	flow velocity of water jet through nozzle	($m \cdot s^{-1}$)
V	volume of ice storage tank	(m^3)
W	total flow of water（Uc +Um）	($m^3 \cdot h^{-1}$)

c_l	specific heat of water	($kJ \cdot kg^{-1} \cdot K^{-1}$)
h_L	latent heat of ice	($kJ \cdot kg^{-1}$)
Δh_w	water level difference fr om normal water level	(m)
IPF	ice packing factor	(%)
N	number of water exchan ges in water region in t he tank	(number h^{-1})
P	arrangement pitch of jet nozzle	(m)
q	cold heat quantity of m elting ice in ice layer	(kW)
q_{load}	thermal load	(kW)
q_{ref}	cooling capacity of an ice generator	(kW)
S	cross-sectional area of tank	(m^2)
T_b	temperature of water flowing out of ice layer	(K)
T_{in}	influent temperature of water for calculation of ice melting	(K)
T_j	temperature of water jet	K (°C)
T_{melt}	melting point of ice	(K)
T_{out}	outlet temperature of ch illed water (intake)	K (°C)
U_c	flow rate of chilled water in ice slurry	($m^3 \cdot h^{-1}$)
U_m	flow rate of water jet	($m^3 \cdot h^{-1}$)
u_j	flow velocity of water jet through nozzle	($m \cdot s^{-1}$)
V	volume of ice storage tank	(m^3)
W	total flow of water（Uc +Um）	($m^3 \cdot h^{-1}$)

Greek letters

ε	porosity of ice layer	(-)
ρ_l	specific gravity of water	($kg \cdot m^{-3}$)
η	utilization rate of stored cold heat (stored ice)	(-)

2. Basic design of chilled water feeder

2.1 Cooling load and conditions of chilled water supply

As a design condition for the chilled water feeder, we show the half-hour cooling load (maximum cooling load) from 0:00 to 24:00 in Figure 1. And Table 1 shows the instantaneous maximum of the design cooling load and the total cooling load for 24 h, as seen in Figure 1, along with the design conditions for the chilled water supply temperature. As shown in Table 1, the instantaneous maximum of the cooling load is 713 kW, and the total cooling load for 24 h is 33.1 GJ. The design condition of the chilled water supply temperature for the cooling load in Figure 1 is 1.0°C (Celsius) or less. In addition, at the maximum load, an increase in the chilled water supply temperature to 2°C within 3 h is permissible.

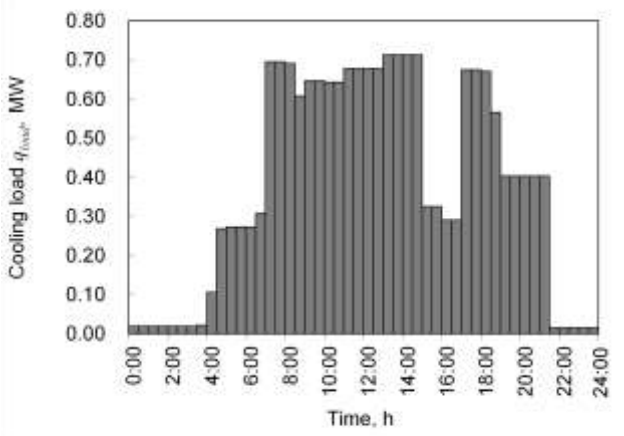

Figure 1. Cooling load pattern (Maximum. dairy cooling load for design)

Table 1. Conditions of Cooling load and design

Max. Cooling load	MW	0.713
Total cooling load	GJ	33.1
Required temperaure of chilled water	℃	1.0 or less within permissible limits of 3 hr @2.0

2.2 Predictive calculation model for chilled water temperature

In the chilled water feeder, the ice storage operation and melting operation are conducted simultaneously. Figure 2 shows a model of an ice storage tank. The ice in the supplied ice slurry increases the region of the ice layer in the tank, and the water in the ice slurry passes through the ice layer and flows into the water region. After the water, whose temperature has been raised by the thermal load, is cooled in the ice layer, it flows into the water region.
To develop a tool for easily predicting the time-dependent change in the chilled water temperature, T_{out}, the followings were assumed:

1) The ice layer is completely separated from the water region.
2) The ice layer is at the freezing-point temperature (0°C).
3) Complete mixing is conducted in the water region.

The equation for the heat storage quantity in the water region utilizes the temperature and flow rate of the water flowing in and out, the temperature of the water region T_{out}, and the

ISBN 978 979 8510 61 8

volume of the water region (the product of the thermal storage utilization rate, η, and tank volume, V) as shown in Equation (1) in Table 2. The cold heat quantity of the melting ice, q, used to obtain the temperature of the water flowing into the water region from the ice layer, T_b, in Equation (1) was formulated using V and the temperature of the influent water, T_{in}, as shown in Equation (2). β_j (melting heat quantity of ice per unit volume/unit temperature difference) in Equation (2) was obtained from Equation (3) (empirical formula). η was obtained from Equation (4). And, as shown in Equation (5), the weighted average efficiency of the temperature of the water jet, T_j, and the temperature of the chilled water in the ice slurry, T_{melt}, was applied to the temperature of the influent water, T_{in}, in Equation (2).

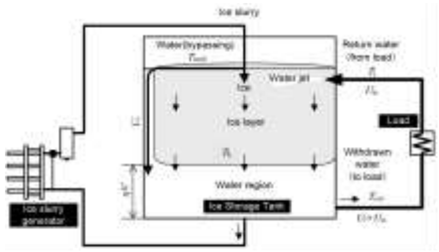

Figure 2. Calculation model of ice storage tank

Table 2. Equation and empirical formula for prediction

$$c_l\rho_l\eta V\frac{dT_{out}}{dt} = c_l\rho_l U_c T_{melt} + c_l\rho_l U_m T_b - c_l\rho_l(U_c + U_m)T_{out} \qquad (1)$$

$$\begin{aligned} q &= c_l\rho_l U_m (T_b - T_j) \qquad (2^{\circ}\mathrm{C/s} \leqq T_{in} < 14^{\circ}\mathrm{C}) \qquad (2)\\ &= \beta_j V (T_{in} - T_{melt}) \end{aligned}$$

$$\begin{aligned} \beta_j &= C_j \cdot g_1(u_j,\eta)\cdot g_2(N)\cdot g_3(P) && (3)\\ g_1(u_j,\eta) &= (0.3 + u_j^{0.1})(1-\eta)^a && (0 \leqq \eta < 1)\\ a &= 0.08(10 - u_j)^{0.1} + 0.2 && (0.2\mathrm{m/s} \leqq u_j < 20\mathrm{m/s})\\ g_2(N) &= \eta N = \eta(W/\eta V) && (0.2\mathrm{h}^{-1} \leqq W/V < 3\mathrm{h}^{-1})\\ g_3(P) &= 1.2 - 0.2P/P_0 \quad ,P_0 = 3m && (1\mathrm{m} \leqq P < 3\mathrm{m}) \end{aligned}$$

$$\eta = \int(q_{load} - q_{ref})dt \,/\{c_l\rho_l V(T_{out} - T_{melt}) + \rho_i h_i V(IPF_0/100)\} \qquad (4)$$

$$T_{in} = U_m \cdot T_j/(U_m + U_c) + U_c \cdot T_{melt}/(U_m + U_c) \qquad (5)$$

2.3 Basic design

As a result of previous experiments and an examination based on numerical analysis[6], it was found that the temperature of the water that is supplied from the tank can be kept below a given temperature if the ice quantity in the tank is controlled to maintain a certain amount or more.

Therefore, we used this calculation tool to predict the time-dependent change in the chilled water temperature in a case where the cooling load in Figure 1 is processed, using parameters that included the various tank capacities, maximum *IPF*, and the refrigerating capacity. Based on the result of the calculations, the conditions that meet the conditions for the chilled water supply temperature in Table 1 were found, and the capacities of refrigerator and ice storage tank were determined. In addition, the number of partitions in an ice generator system was examined.

2.4 Design capacity of refrigerator and ice storage tank

As a result of a calculation in a case where the capacity of ice storage tank is 68.3 m^3 and the maximum *IPF* (IPF_0) is 45%, the maximum value of the chilled water supply temperature and

ISBN 978 979 8510 61 8

the minimum *IPF* in the tank for various refrigerators are shown in Figure 3. It was found that the capacity needed by a refrigerator to maintain the minimum *IPF* at more than 20% was more than 510 kW, while that needed to maintain the maximum value of the chilled water supply temperature at less than approximately 1°C (1.0 ± 0.1°C) was more than 490 kW, as shown in the figure. This refrigerator capacity is approximately 70% of the maximum cooling load in Table 1. In addition, it was found that the refrigerator capacity should be more than 415 kW to maintain the maximum value of the chilled water supply temperature at 2°C or less, which is the permitted condition.

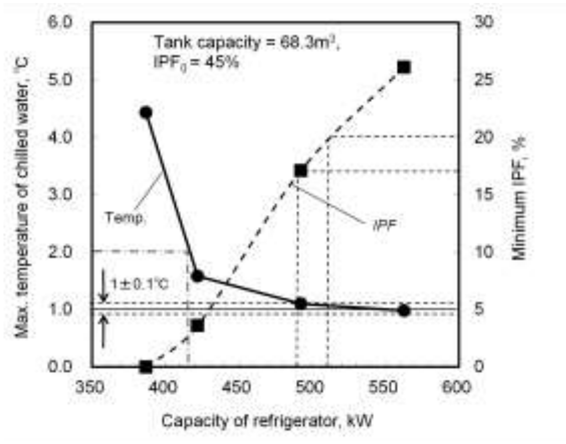

Figure 3. Correlation of refrigerator capacity with

As a result of a calculation in a case where the capacity of refrigerator is 490 kW and the maximum *IPF* (IPF_0) is 45%, the maximum value of the chilled water supply temperature and the minimum *IPF* in the tank for various tank capacities are shown in Figure 4. It was found that the capacity needed by an ice storage tank to maintain the minimum IPF at more than 20% was more than 76.0 m^3, while that needed to maintain the maximum value of the chilled water supply temperature at less than approximately 1°C (1.0 ± 0.1°C) was more than 68.3 m^3, as shown in the figure. In addition, it was found that the tank capacity should be more than 45 m^3 to maintain the maximum value of the chilled water supply temperature at 2°C or less, which is the permitted condition.

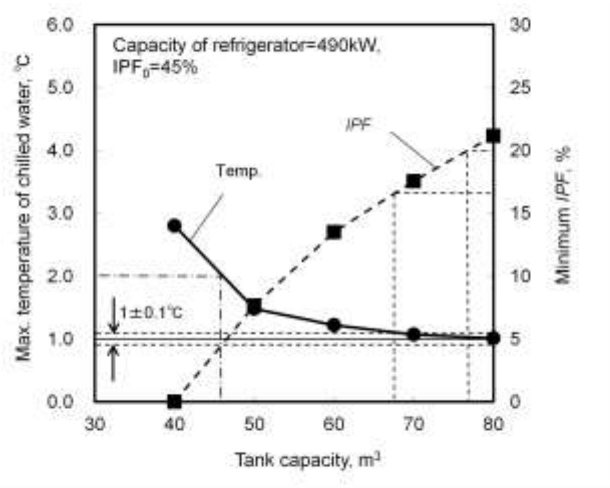

Figure 4. Correlation of tank capacity with maximum temperature of chilled water and minimum *IPF*

The equipment capacities (the capacities of an ice storage tank, refrigerator, and super-cooler), which were determined based on the above calculation results and the permissible value of chilled water supply temperature, are shown in Table 3.

Table 3 Determined capacities of facility

Ice storage tank	Capacity	m^3	68.3
	Size	m	7×3×4H (water level : 3.25)
	Max. IPF	%	45
Refrigerator		kW	492 (246 ×2)
Super-cooler		kW	492 (246 ×2)

2.4 Design of number of partitions in ice generator system

In a food factory that operates 24 h a day, a problem with a refrigerator is fatal to the operation of the factory. In the case of a thermal storage system, the cold energy release caused by the melting operation can prevent the cooling process from stopping suddenly even if an unexpected failure occurs. Thus, in order to evaluate the function of an ice storage tank as emergency equipment, we simulated the processing of a thermal load using only the cold energy release from the melting operation, under the assumption of an ice generator failure and a variable thermal load, similar shape to that shown in Figure 1. Figure 5 shows the correlation of the retention time when the chilled water temperature can be kept at 1.0°C or less and the load factor (daily load / 33.1 GJ) parameters, which include the number of partitions in ice generator system.

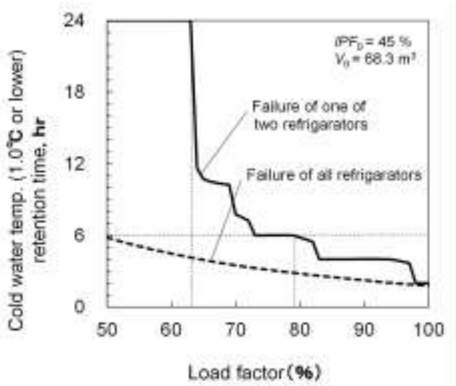

Figure 5. Correlation of ice generator Load factor with maximum temperatur e of chilled water

As shown in Figure 5, a smaller load factor will produce a longer period when a chilled water temperature of 1.0°C or less can be maintained. In a case where the load factor is 100%, the temperature of the chilled water can be kept at 1°C or less for up to 1.8 h. Even if all the refrigerators fail, the temperature of the chilled water can be kept at 1.0°C or less for approximately 6 h by operating at a load factor of approximately 50%, or limiting the processing capacity of a food product plant to approximately 50%. In addition, if one of two refrigerators fails, the temperature of the chilled water can be kept at 1.0°C or less for 6 h by operating at a load factor of 73~79%, and for 24 h by reducing the load factor to 63% or less.
As seen above, an ice storage tank with an *IPF* of 45% was judged to be capable of functioning as emergency equipment. Based on this, the number of partitions was determined to be 2, and the capacity of each ice generator was determined to be 246 kW as shown in Table 3.

2.5 Operation control

Here, we will explain the method used to determine the cold heat storage quantity (fully

ISBN 978 979 8510 61 8

accumulated state) and the operation control method for the makeup water.

2.6 Control in fully accumulated state (management and control of heat storage quantity)

Because a cooling load is being continuously applied 24 h a day in the dairy product plant where this equipment was introduced, the ice storage operation and melting operation (chilled water supply operation) are conducted almost simultaneously all day long. If the load is smaller than the cooling capacity of the ice generator, the ice storage quantity increases, and if the load is larger than that of the ice generator, the ice storage quantity decreases. If the ice storage quantity (cold heat storage quantity), which increases or decreases, is measured only by the addition of the heat quantity, calculation errors accumulate over time.
Thus, we developed a method to determine the fully accumulated state based on the expanding height of the ice layer and adopted it for the actual control. The control logic is shown in Figure 6. The ice making operation stops when the expanding height of the ice layer reaches a given level of 450 mm (because it is judged to be at a fully accumulated state). At this time, the value of the calculated cold heat storage quantity is reset (to *IPF* = 45%). This enables the regular resetting of the error in the calculated cold storage heat quantity.
The quantity of ice that has been consumed by load processing after reaching the fully accumulated state is obtained by calculating the heat quantity, and the ice making operation starts when *IPF* decreases to 43%.

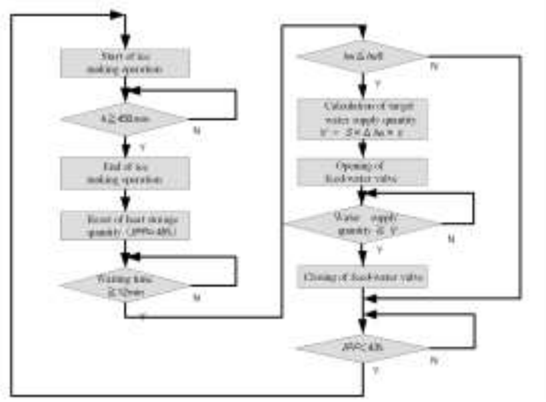

Fig.6 Logic of operation control

2.7 Control of makeup water

When the water retained in the piping system is discharged by cleaning the heat exchanger on the food processing side, the quantity of water in the ice storage tank decreases. Thus, in order to maintain a certain quantity of water in the tank at all times, a control operation detects the required quantity of makeup water and immediately supplies water during the ice storage system operation when required.

Thus, the makeup water feed control logic shown in Figure 6 was developed and adopted in this system. In the ice storage condition where *IPF* is more than 20%, the ice layer does not float on the water but attaches to the bottom of the tank. Therefore, the reduction of water in the tank can be detected by observing the difference between the water level in the tank and the normal water level. However, as the tank is filled with a sherbet-like ice layer, it is necessary to calculate the water supply quantity by multiplying the water level difference by the porosity of the ice layer (ε = 0.7). Prior to the design, tests were conducted to measure the porosities of the ice layers in variable thermal storage tanks with different capacities, and it was found that they had an almost constant value (ε = 0.7) regardless of the capacity of the

ISBN 978 979 8510 61 8

thermal storage tank. In addition, because the water that flows in the ice layer does not stop immediately after reaching the fully accumulated state, the water level changes even after ice water stops being supplied to the thermal storage tank. Thus, we waited 12 min after it reached the fully accumulated state, and measured the water level in the tank using a level gauge after the water flowing in the ice layer dropped and the water level became stable. We decided to supply the tank with a volume of water based on the calculation of Equation (6).

$$V = \Delta h_{w} \times S \times \varepsilon \tag{6}$$

In this way, by starting the water supply after determining the target water supply quantity based on the water level difference, the given water supply quantity can be ensured, even if the ice making operation starts during the water supply and the water level changes.

3. Outline of chilled water feeder

3.1 Flow of feeder

Figure 7 shows the actual flow of the chilled water feeder that was introduced in the dairy product plant. This equipment included an ice storage tank (1), heat exchanger that heats the water (0°C) taken from the tank to 0.5°C (pre-heater) (2), heat exchanger that generates supercooled water at -2°C (super-cooler) (3), unit where the super cooled condition of the water is released and fine ice is generated (releaser) (4), brine refrigerator (5), two ice generators composed of pumps (6) and other components (7), and chilled water pumps (8). By making the ice storage tank a cold-energy buffer, the capacity of the refrigerator decreases, highly efficient operation becomes possible, and low-temperature chilled water is stably supplied from the ice storage tank.

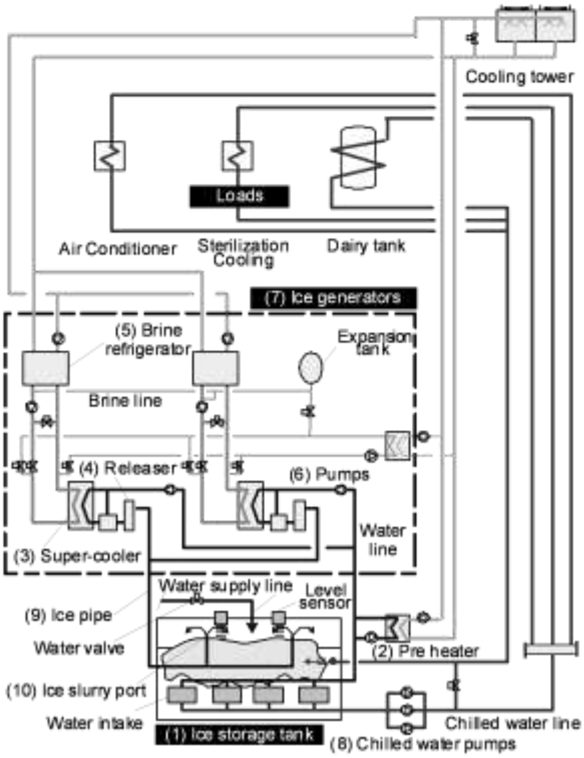

Figure 7. Schematic diagram of chilled water feeder adopted with dynamic-type ice storage system using super cooling water

In the ice storage operation, the ice slurry that is generated in the ice generator is pumped

ISBN 978 979 8510 61 8

through the ice pipe (9) to the ice storage tank (1), and is spouted through the two ice slurry ports (10), which are installed near the water surface in the tank. The ice slurry that is supplied to the tank is separated into ice and water by the specific gravity difference. The water at the bottom of the tank is preheated by the pre-heater (2), and then is supplied to the ice generator (7) again. The sherbet-like ice layer is stored in the tank using this water circulation.
In the chilled water supply operation, the low-temperature water at the bottom of the ice storage tank is sent to the heat exchanger on the cooling load side by the chilled water pump. After the load processing causes its temperature to rise, the water returns to the tank, is spouted through the jet nozzle into the tank, and melts the sherbet-like ice (dynamic ice).

3.2 Control in fully accumulated state and control of makeup water

At the top of the tank, as shown in Figure 7, ultrasonic level sensors are set in order to control the fully accumulated state, and the expansion height of the ice layer is measured. In addition, to control the makeup water, an automatic valve is installed in the water supply line. This valve opens after reaching the fully accumulated state, and the valve closes at the point when the integrated value of a flowmeter reaches the target value.

4. Operation results with actual equipment

4.1 Change in cooling load and chilled water temperature

Figure 8 shows an example of the cooling load q_{load} and the time-dependent changes in the chilled water supply temperature, T_{out}, and *IPF* in the ice storage tank. The interval for the measured data was 1 min.
As shown in Figure 8, at approximately 10:00 and 16:00, instantaneous maximum load that were more than 1.5 times larger than the design maximum value occurred. In addition, the integrated value for 24 h was 34.9 GJ, which was 105% of the design maximum integrated value, 33.1 GJ. Because the cooling load, q_{load}, was extremely small from 0:00 to 7:00, the chilled water supply temperature, T_{out}, stayed at a low value of approximately 0.1°C during most of this time. When q_{load} increased sharply at around 4:00, T_{out} rose to approximately 0.6°C. At around 9:00, T_{out} temporarily rose to 1.2°C with a sharp increase in q_{load} until q_{load} exceeded the design maximum load. However, after that, T_{out} could be maintained at 1.0°C or less at all times.

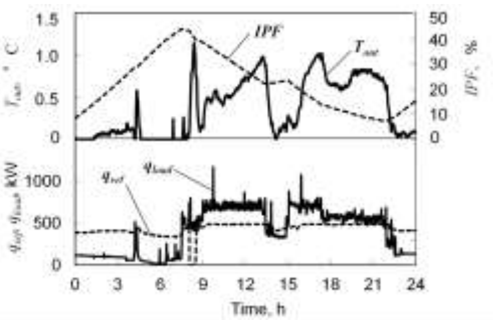

Fig.8 Operation results for cooling load, chilled water temperature, *IPF* and q_{ref} on actual cooling load

4.2 Control in fully accumulated state

As shown in above Figure 8, from 0:00 to 7:30, *IPF* increased monotonously as the cooling capacity of the ice generator, q_{ref}, surpassed the cooling load, q_{load}. When the ultrasonic level sensor sensed that the expansion height of the ice layer reached the set value at 8:00, the condition was judged to be the fully accumulated state (*IPF* = 45%), and the ice generation

ISBN 978 979 8510 61 8

operation stopped (q_{ref} = 0). Figure 9 shows the appearance inside the ice storage tank. After that, *IPF* was decreased by the cooling load processing, and fell below the predetermined start condition (*IPF* = 43%) for the ice making operation at 8:30, at which time the ice making operation was restarted. After that, as q_{load} surpassed the cooling capacity of the ice generator, *IPF* decreased. Although *IPF* dropped to approximately 7% by 22:00, when q_{load} dropped sharply, it increased to approximately 13% at 24:00, and it surpassed the *IPF* that occurred 24 h previously (0:00 in the figure).
In this way, it was found that when the expanding height of the ice layer was used to determine the fully accumulated state, the reset of the calculated ice storage quantity based on the ice storage quantity (*IPF*) in the fully accumulated state, and the start of the ice making operation based on the calculated ice storage quantity were properly conducted in the actual operation. Moreover, as a result of the start-stop control of the ice making operation, it was found that an ice storage quantity that could stably provide chilled water at a given temperature against the cooling load, which continued to occur for 24 h, could be maintained.

Figure 9. Appearance inside ice storage tank at fully accumulated state (*IPF* = 45%)

5. Conclusion

We implemented the basic design of a chilled water feeder for a food plant using a dynamic-type ice storage system that utilized the supercooling phenomenon of water. Moreover, we evaluated the method used for the basic design based on the operation results for an actual load using the actual equipment. The following conclusions could be reached:

1) By conducting a predictive calculation of the chilled water temperature based on the design cooling load, the capacity of a refrigerator and that of an ice storage tank for supplying low-temperature chilled water at approximately 1°C for the food process were determined. In addition, the number of partitions for an ice generator system was determined from the viewpoint of the continuity of the process with an ice generator problem.
2) As a method of operation control, we developed the control logic for the fully accumulated state and the makeup water control, and adopted these in the execution design. The operation of the actual equipment demonstrated that this design operated without problems.
3) It was found that the capacity of the refrigerator and that of the ice storage tank, which had been determined in the design examination, could supply low-temperature chilled water at approximately 1°C through the actual operation.

6. References

[1] Tanino, M., Shiraishi, H., Hayashi, T., Okonogi, T., Okada, T., Ice Storage System with Supercooled Water., SHASE. 1990;64(7):51. (in Japanese)

[2] Kozawa Y., Tanino M., Ice-water Two-phase Flow Behavior in Ice Heat Storage Systems, Proc. of 1st workshop on ice slurries of IIR. 1999:146-156.

[3] Mito, D., Kozawa, Y., Tanino, M., Inada, T., Development of Active Control Technology relating to Supercooling Release of Water. JSRAE. 2000; 17(2):191. (in Japanese)

[4] Tanino M., Kozawa Y., Mito D., Inada T., Development of Active Control Method for Supercooling Releasing of Water, Proc. of the 2nd workshop on ice slurries of IIR. 2000:127-139.

[5] Mito D., Mikami Y., Tanino M., Kozawa Y., A New Ice-Slurry Generator by using Actively Thermal-hydrauling Controlling both Supercooling and Releasing of Water, Proc. of 5th workshop on ice slurries of IIR. 2002:185-196.

[6] Mito D., Tanino M., Kozawa Y., Okamura. A., Application of a Dynamic-type Ice Storage System to the Intermittent Cooling Process in the Food Industry, Proc. of 4th workshop on ice slurries of IIR. 2001:105-114.

[7] Mito, D., Man'o, T., Hongo, M., Wakasa, K., Application of Dynamic-type Ice Storage System to Food Cooling Process–Introduction of Chilled Water Feeder in a Dairy Product Plant. Proc. of 2012 JSRAE Annual Conference. 2012: 479-482. (in Japanese)